NASA/TM—2007-214954 ISCORMA–4–510

Vibration Transmission Through Bearings With Application to Gearboxes

David P. Fleming
Glenn Research Center, Cleveland, Ohio

July 2007

NASA STI Program . . . in Profile

Since its founding, NASA has been dedicated to the advancement of aeronautics and space science. The NASA Scientific and Technical Information (STI) program plays a key part in helping NASA maintain this important role.

The NASA STI Program operates under the auspices of the Agency Chief Information Officer. It collects, organizes, provides for archiving, and disseminates NASA's STI. The NASA STI program provides access to the NASA Aeronautics and Space Database and its public interface, the NASA Technical Reports Server, thus providing one of the largest collections of aeronautical and space science STI in the world. Results are published in both non-NASA channels and by NASA in the NASA STI Report Series, which includes the following report types:

- TECHNICAL PUBLICATION. Reports of completed research or a major significant phase of research that present the results of NASA programs and include extensive data or theoretical analysis. Includes compilations of significant scientific and technical data and information deemed to be of continuing reference value. NASA counterpart of peer-reviewed formal professional papers but has less stringent limitations on manuscript length and extent of graphic presentations.

- TECHNICAL MEMORANDUM. Scientific and technical findings that are preliminary or of specialized interest, e.g., quick release reports, working papers, and bibliographies that contain minimal annotation. Does not contain extensive analysis.

- CONTRACTOR REPORT. Scientific and technical findings by NASA-sponsored contractors and grantees.

- CONFERENCE PUBLICATION. Collected papers from scientific and technical conferences, symposia, seminars, or other meetings sponsored or cosponsored by NASA.

- SPECIAL PUBLICATION. Scientific, technical, or historical information from NASA programs, projects, and missions, often concerned with subjects having substantial public interest.

- TECHNICAL TRANSLATION. English-language translations of foreign scientific and technical material pertinent to NASA's mission.

Specialized services also include creating custom thesauri, building customized databases, organizing and publishing research results.

For more information about the NASA STI program, see the following:

- Access the NASA STI program home page at *http://www.sti.nasa.gov*

- E-mail your question via the Internet to *help@sti.nasa.gov*

- Fax your question to the NASA STI Help Desk at 301–621–0134

- Telephone the NASA STI Help Desk at 301–621–0390

- Write to:
  NASA Center for AeroSpace Information (CASI)
  7115 Standard Drive
  Hanover, MD 21076–1320

NASA/TM—2007-214954 ISCORMA–4–510

Vibration Transmission Through Bearings With Application to Gearboxes

David P. Fleming
Glenn Research Center, Cleveland, Ohio

Prepared for the
Fourth Biennial International Symposium on Stability Control of Rotating Machinery (ISCORMA–4)
sponsored by the Bently Pressurized Bearing Company
Calgary, Alberta, Canada, August 27–31, 2007

National Aeronautics and
Space Administration

Glenn Research Center
Cleveland, Ohio 44135

July 2007

Acknowledgments

Thanks are due to Dr. Florin Dimofte of the University of Toledo for supplying the wave bearing data.

Level of Review: This material has been technically reviewed by technical management.

Available from

NASA Center for Aerospace Information
7115 Standard Drive
Hanover, MD 21076–1320

National Technical Information Service
5285 Port Royal Road
Springfield, VA 22161

Available electronically at http://gltrs.grc.nasa.gov

Vibration Transmission Through Bearings With Application to Gearboxes

David P. Fleming
National Aeronautics and Space Administration
Glenn Research Center
Cleveland, Ohio 44135

Abstract

Cabin noise has become a major concern to manufacturers and users of helicopters. Gear noise is the largest part of this unwanted sound. The crucial noise path is generally considered to be from the gears through the gear-supporting shafts and bearings into the gearbox case, and from there either through the gearbox mounts or the surrounding air to the helicopter cabin. If the noise, that is, the gear and shaft vibration, can be prevented from traveling through the gearbox bearings, then the noise cannot make its way into the helicopter cabin. Thus the vibration-transmitting properties of bearings are of paramount importance. This paper surveys the literature concerning evaluation of properties for the types of bearings used in helicopter gearboxes. A simple model is proposed to evaluate vibration transmission, using measured or calculated bearing stiffness and damping. Less-commonly used types of gearbox bearings (e.g., fluid film) are evaluated for their potential in reducing vibration transmission.

Introduction

Early geared transmissions operated at low speed where little attention had to be given to noise control. Examples are the many traditional Dutch windmills built to pump water or cut wood. A particular example is the wind-powered sawmill “De Rat” (ref. 1) located in IJlst, the Netherlands (fig. 1). In this machine, originally built in the seventeenth century, the four-blade wind turbine is on a near-horizontal shaft, and turns at a maximum speed of 20 rpm. A gearset (fig. 2) transfers motion to a vertical intermediate shaft. A second speed-increasing gearset turns a horizontal three-throw crankshaft, which in turn drives three banks of reciprocating saw blades at a maximum speed of 40 strokes per minute. Much of the massive windmill structure, including the gear teeth, is wood. Thus vibration attenuation is substantial.

In contrast to these windmills, modern helicopter transmissions may operate with input speeds of 20,000 rpm. In addition, helicopter designers strive to keep weight to a minimum. High speed and low weight, with resulting high power density, make noise a major problem. Much of the noise originates in the gear meshes. The noise travels by various paths into the helicopter crew and passenger compartments where it contributes to fatigue and hearing loss.

One of the significant noise paths is from the gear mesh, through the transmission shafts and bearings, into the transmission case. From there it may travel through the transmission mounts into the airframe structure, or be radiated through the air. If noise can be prevented from traveling through the transmission bearings, significant quieting of the helicopter will occur. One way proposed to accomplish this is to replace the normally-used rolling-element transmission bearings, which have very little damping, with fluid film bearings. The much higher damping in fluid film bearings is expected to significantly reduce noise.

This paper proposes a simple model for noise (i.e., vibration) transfer through transmission bearings. The effect of bearing stiffness and damping will be explored.

Figure 1.—Wind-powered sawmill.
(Photograph by the author.)

Figure 2.—Wooden gears in windmill.
(Photograph by the author.)

Background

An extensive review of gear housing dynamics and acoustics literature was published by Lim and Singh in 1989 (ref. 2). In this review, they note that gearbox vibration and noise are due to *transmission error*, which is a measure of the accuracy with which the driven gear follows the driver gear. Transmission error is due to imperfection in gear manufacture, and also to elastic deformation of the gear teeth as they move through mesh. Transmission designers strive to minimize the error, but it can never be eliminated. The vibration frequency of transmission error is the gear mesh frequency and multiples thereof, and also sidebands of these frequencies; that is, the mesh frequencies plus or minus the gearbox shaft frequencies.

Lim and Singh followed their literature survey with a five-part study of vibration transmission through rolling element bearings (refs. 3 to 7). In these papers, they note that bearing translational properties alone do not adequately explain how vibration is transferred from the rotating shaft to the transmission case. They propose use of a 6×6 matrix of bearing stiffness (including cross-coupling coefficients), which includes radial, angular, and axial motion. They calculate stiffness matrices for one example each of ball and roller bearings.

Liew (ref. 8) (a student of Lim), and Liew and Lim (ref. 9) calculated the time-varying bearing stiffness as the balls orbit in their track. The authors termed the new analysis a "slight improvement" over earlier work.

Joshi (ref. 10) (another student of Lim) refined the elastic contact analysis to include deformation of the raceway as well as the ball. He also calculated the time-varying bearing stiffness due to ball orbiting.

Royston and Basdogan (ref. 11) carried out an analysis for spherical rolling element bearings, also including the cross-coupling terms. In this case, the bearing has no angular stiffness.

Bearing damping was not considered in any of these analyses, but it is usually assumed to be very low (refs. 12 and 13).

Fleming (ref. 14) conducted a study in which the effects of bearing translational stiffness and damping on transmission error were investigated. In this case, static transmission error was given; proper choice of bearing properties reduced dynamic transmission error up to 10 dB. Transmission case flexibility was not taken into account.

Analytical Model

A geared transmission consists of at least two geared shafts mounted in bearings. The transmission case is a three-dimensional structure not readily amenable to a simple representation; usually a finite-element model is employed. For the present study, an extremely simple model is proposed as shown in figure 3.

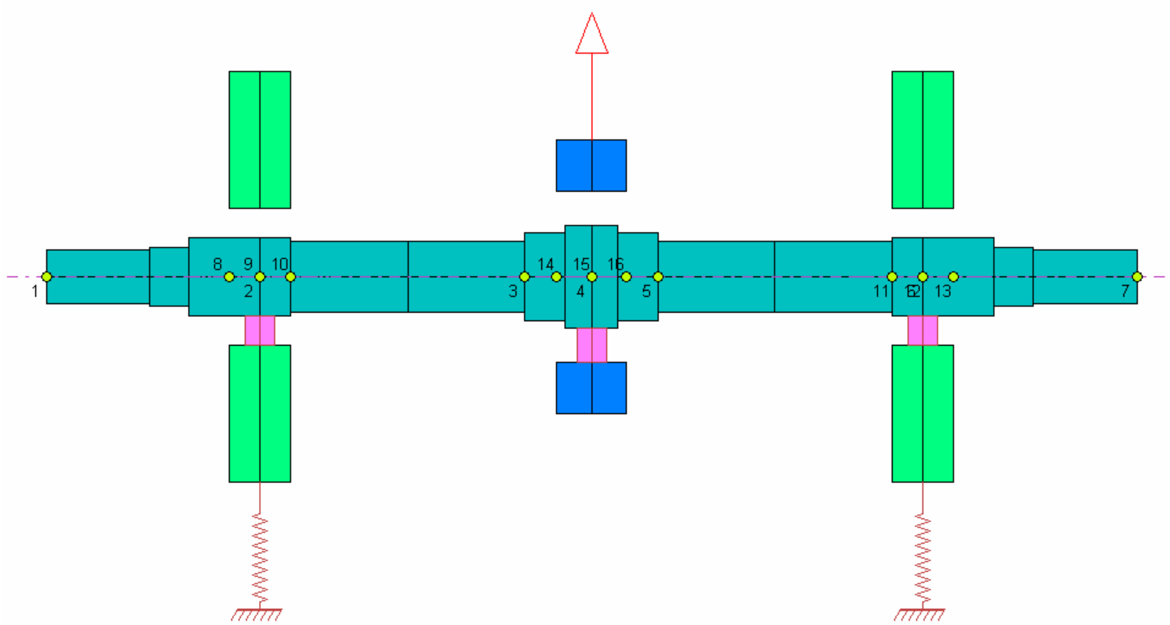

Figure 3.—Analytical model.

The model stems from the gear noise rig at NASA Glenn Research Center. This rig has two counter-rotating shafts, each with a 28-tooth gear. Noise is generated by the gear mesh due to transmission error; it manifests itself as a periodic force between engaging gear teeth. This force may be resolved into a radial force on the shafts plus shaft torque; it is the radial force that is felt by the bearings and is analyzed here. In the model, only one of the shafts is considered. Rather than a periodic radial force, an unbalanced ring (shown in blue in figure 3) at the shaft midspan applies a rotating load to the shaft; the unbalance force is represented in figure 3 by the red arrow. The ring is assumed to be very light, so that it does not affect the model dynamics except through the unbalance force. It is connected to the shaft by very stiff bearings (1.75 GN/m). Having the ring separate from the shaft allows nonsynchronous excitation frequencies to be used; for the present work, mesh frequency (28 times shaft speed) and twice mesh frequency (56 times shaft speed) were used. Use of a rotating unbalance load is obviously different than the periodic unidirectional force assumed in an actual transmission; however, the technique should be useful in evaluating the effect on vibration of different bearings.

The shaft was modeled after that in the noise rig; however, it was made symmetric end-to-end for simplicity in interpreting results. Two bearings with a span of 245 mm support the shaft. The small numerals along the shaft centerline in figure 3 are the station numbers of the rotordynamic model.

The portions of the transmission case near the bearings are modeled by nonrotating shafts (shown as green rings in fig. 3). They are connected to the remainder of the case (assumed to be rigid and unmoving) by the bearings shown as the red sawtooth lines, and to the shaft by bearings shown in pink. The shaft and case-simulating rings are taken to be steel. At this point a finite element model of the actual transmission case has not been employed; the hollow shafts representing the case were merely sized to “look” right.

Two well-known rotordynamics codes, DyRoBeS (ref. 15) and ARDS (ref. 16), were used to carry out the analysis. DyRoBeS is a Windows development of ARDS and is the more user-friendly code. However, availability of the ARDS source code allowed output to be tailored to the present application.

Bearings

As mentioned, Lim and Singh calculated the stiffness of a ball bearing (ref. 3). The bearing size was not given explicitly, but it appears to be a 25 mm bore, light series bearing. This is close to a size that could be used in the noise rig, so the direct radial stiffness coefficient and direct angular stiffness coefficient calculated in (ref. 3) were used in the analysis. Cross-coupled coefficients were not used, nor were axial coefficients since it was assumed that the bearings do not carry any thrust load.

The other bearing type that was simulated was the *wave bearing* (ref. 17). This bearing is similar to a full circular journal bearing, but there is a variation in clearance around the circumference that greatly reduces bearing instability tendencies. In common with other fluid film bearings, and in contrast to rolling-element bearings, the wave bearing has substantial damping. Wave bearings were specifically designed for the noise rig; they have a 32 mm diameter and are 20 mm long.

There is some ambiguity in choosing stiffness coefficients for the wave bearing, since fluid film bearing stiffness varies with both load and speed. The load that the bearing will experience in the noise rig is not precisely known, and the rig will operate over a range of speeds. Values of stiffness were chosen for a moderately loaded bearing corresponding to an eccentricity ratio of 0.34, and a speed of 4000 rpm (ref. 18). Angular stiffness coefficients for the wave bearing were not supplied; they were approximated by assuming that the bearing film pressure profile is parabolic in the axial direction. Thus,

$$K_\theta = K_x L^2/24$$

where L is the bearing length; angular damping was calculated similarly. Bearing properties are shown in Table 1. The case stiffnesses were arbitrarily chosen to be the same as those of the ball bearing. Ball bearing damping was estimated from (refs. 12 and 13); the radial damping is much less than that of the wave bearing, but the angular damping is comparable.

TABLE 1.—BEARING AND CASE PROPERTIES

	Radial stiffness K_x, MN/m	Angular stiffness K_θ, N m/rad	Radial damping B_x, kN s/m	Angular damping B_θ, N m s/rad
Ball bearing	175	15	3.5	2.3
Wave bearing	46.6	0.78	144	2.4
Case	175	15	3.5	2.3

Results

For all results presented herein, excitation was applied to the shaft by means of 7.2 g cm imbalance on the balance ring. Since a linear system has been assumed, the amount of imbalance is immaterial to understanding the results; the results scale linearly. Figures 4 and 5 show the “case” radial amplitude and angular amplitude (slope) respectively for excitation at 28 times shaft speed (corresponding to the mesh frequency of the 28-tooth gears in the NASA Glenn noise rig). The angular vibration occurs because of shaft bending and the angular bearing stiffness and damping. It is posited as a significant means of noise transmission into the atmosphere surrounding the transmission. Both figures exhibit resonance peaks for both bearing types within the speed range plotted. The lower stiffness and higher damping of the wave bearing do not seem to confer an overall advantage regarding noise propagation.

A further check on the efficacy of increased damping is to compare shaft amplitudes at the midspan and bearing location for ball and wave bearings. This is done in figures 6 and 7, respectively. In figure 6, midspan vibration amplitudes at low speed (below 4000 rpm) are similar for both bearings. At higher speed with the wave bearing, vibration builds to a sharp peak around 6500 rpm; the ball bearing produces a much lower peak at a lower speed. At the bearing location, figure 7, amplitude is lower for the wave bearing at low speed, but higher above 6000 rpm. Thus the wave bearing has not produced lower overall shaft vibration.

Insight into this unexpected behavior may be had by considering that the vibration excitation is at much higher frequency than shaft speed; 28 and 56 times. The force produced by the damping varies as the term ωB, where ω is the excitation frequency. This force is applied similar to that from the bearing stiffness. Thus for a shaft speed of 6000 rpm, for example, the vibration frequency is $28 \times 6000 \times \pi/30 = 17593$ rad/s, and for the wave bearing damping of 144 kN s/m the effective stiffness ωB due to damping is some 2500 MN/m. This far overshadows the wave bearing stiffness of 46.6 MN/m and the case stiffness of 175 MN/m. The shaft and case thus move in unison, with virtually no relative motion between them; note the similarity of the wave bearing curves in figures 4 and 7.

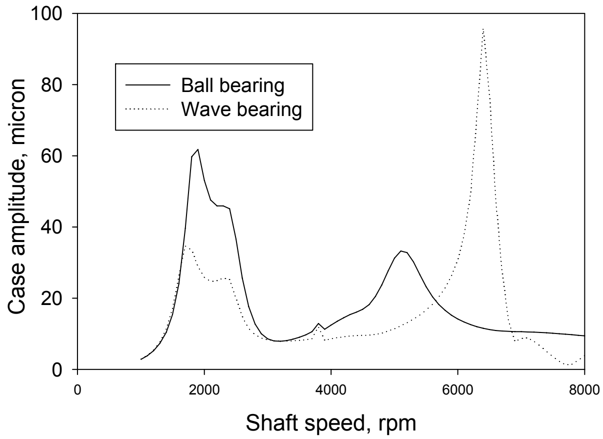

Figure 4.—Radial vibration amplitude of transmission case for 28× excitation.

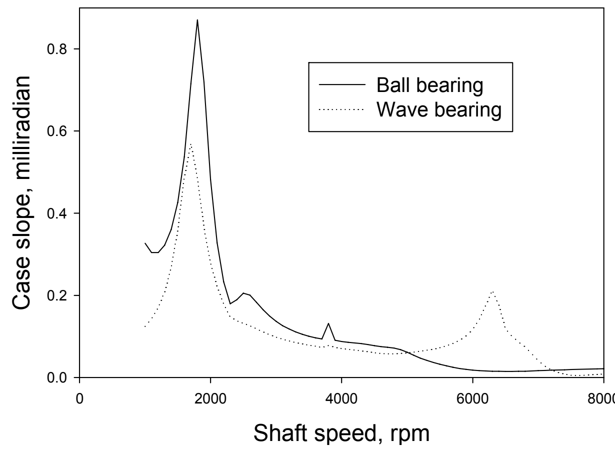

Figure 5.—Angular vibration of transmission case for 28× excitation.

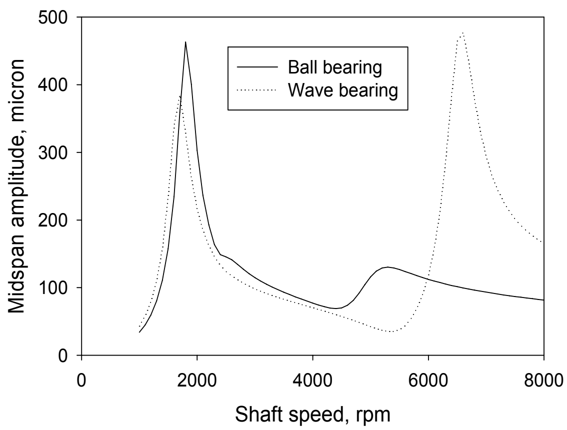

Figure 6.—Rotor midspan amplitude; 28× excitation.

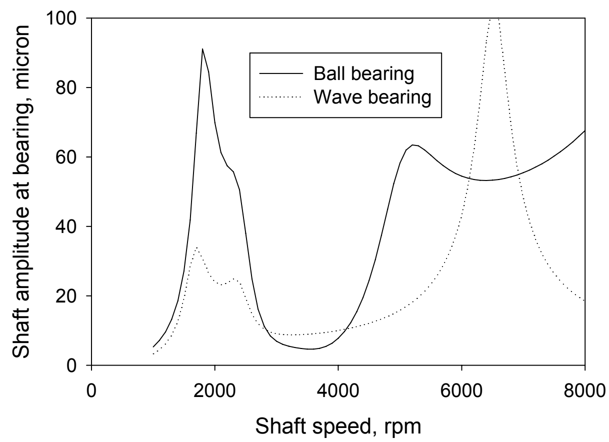

Figure 7.—Shaft amplitude at bearing location; 28× excitation.

Contrary to the results shown above, experimental data for rotors supported on wave bearings indicate that high frequency vibration is effectively damped (ref. 18). Apparently wave bearing damping drops at high frequency in a manner that is not accounted for in the data used herein.

Evidence that the high effective stiffness ωB is responsible for large case amplitudes at high excitation frequencies is in figures 8 and 9. In these figures, the wave bearing damping used in the analysis was cut to one-tenth of its earlier value. Figure 8 shows that the radial vibration peak occurring near 6500 rpm nearly disappears with lower damping. On the other hand, the peak near 2000 rpm has become higher. Figure 9 shows that lower damping markedly lowers angular vibration across the speed range.

Lastly, response to excitation at twice mesh frequency will be presented. Case radial and angular vibration are shown in figures 10 and 11, respectively. These figures, for speeds below 4000 rpm, appear very similar to figures 8 and 9 with the horizontal axis compressed. This is not surprising, as similar vibration modes would be excited at similar frequencies, whether at 28 or 56 times shaft speed (the vibration is not identical because of gyroscopic effects). Beyond 4000 rpm, one more peak appears near 5500 rpm in the curve for the wave bearing with original damping. Two very low peaks occur in angular vibration (fig. 11), near 5500 and 7500 rpm. In both figures, reducing the damping lowers the vibration over most of the speed range, as was the case for 28× excitation.

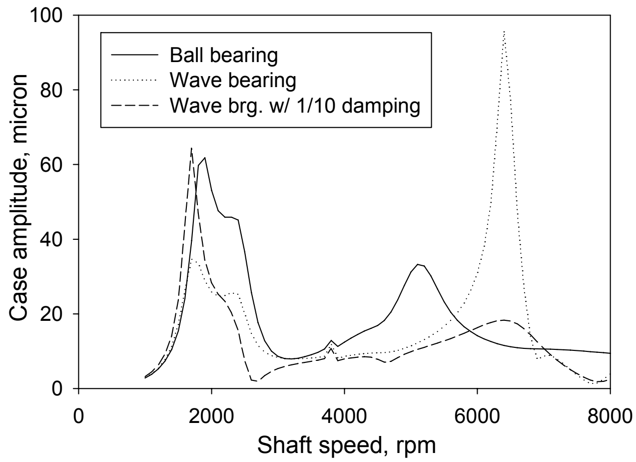

Figure 8.—Radial vibration of transmission case; ball, wave, and wave bearing with reduced damping; 28× excitation.

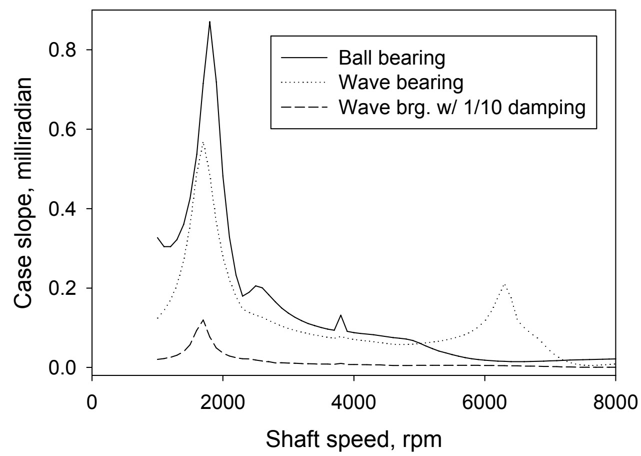

Figure 9.—Angular vibration of transmission case; ball, wave, and wave bearing with reduced damping; 28× excitation.

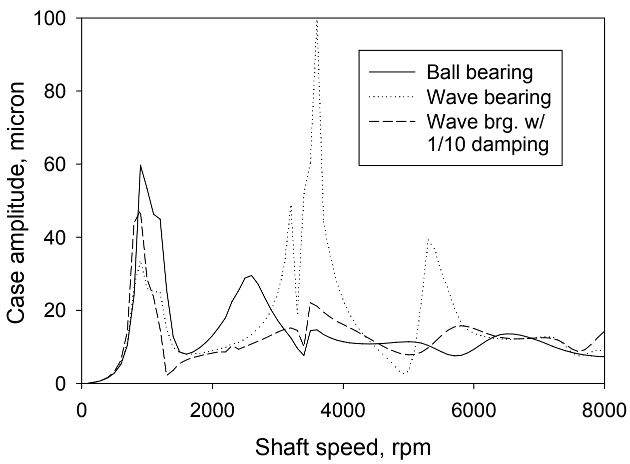

Figure 10.—Radial vibration of case; 56× excitation.

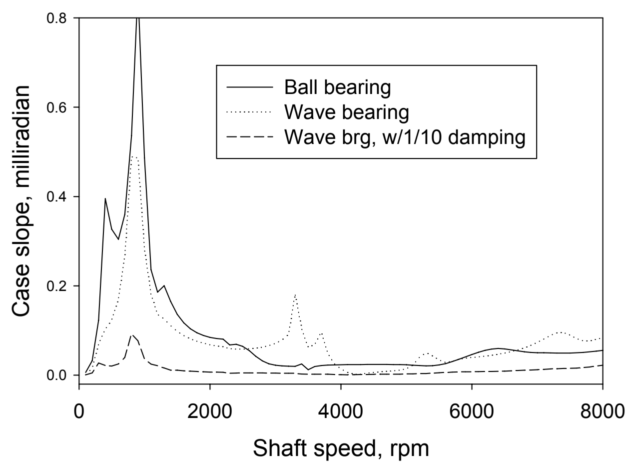

Figure 11.—Angular vibration of case; 56× excitation.

Conclusions

Some of the literature dealing with vibration transfer through rolling element bearings has been discussed in which analyses were done to calculate the stiffness of these bearings. Bearing damping was not considered in these calculations; it is acknowledged to be very low in rolling bearings.

A novel analytical model was described to enable easy calculation of vibration transfer through different bearings once bearing stiffness and damping are known. The model was used in conjunction with calculated ball bearing data, and also for data pertaining to a fluid film wave bearing which has much higher damping than a ball bearing. For the most part, this high damping was not beneficial in reducing vibration transfer at the high frequencies (28 and 56 times shaft speed) involved. Previous experiments indicate that wave bearings actually do reduce high-frequency vibration; however, appropriate damping coefficients were not available for evaluation herein.

References

1. Houtzaagmolen De Rat. http://www.zwfriesland.nl/derat/ (in Dutch)
2. Lim, T. C., and Singh, R. (1989): “A Review of Gear Housing Dynamics and Acoustics Literature,” NASA CR–185148.
3. Lim, T.C., and Singh, R. (1990): “Vibration Transmission through Rolling Element Bearings. Part I: Bearing Stiffness Formulation,” *J. of Sound and Vibration,* vol. 139, no. 2, pp. 179–199.
4. Lim, T.C., and Singh, R. (1990): “Vibration Transmission through Rolling Element Bearings. Part II: System Studies,” *J. of Sound and Vibration,* vol. 139, no. 2, pp. 201–225.
5. Lim, T.C., and Singh, R. (1991): “Vibration Transmission through Rolling Element Bearings. Part III: Geared Rotor System Studies,” *J. of Sound and Vibration,* vol. 151, no. 1, pp. 31–54.
6. Lim, T.C., and Singh, R. (1992): “Vibration Transmission through Rolling Element Bearings. Part IV: Statistical Energy Analysis,” *J. of Sound and Vibration*, vol. 153 no. 1, pp. 37–50.
7. Lim, T.C., and Singh, R. (1994): “Vibration Transmission through Rolling Element Bearings. Part V: Effect of Distributed Contact Load on Roller Bearing Stiffness Matrix,” *J. of Sound and Vibration*, vol. 169, no. 4, pp. 547–553
8. Liew, Hoon-Voon (2002): “Analysis of Time-Varying Rolling Element Bearing Characteristics,” MS Thesis, University of Alabama.
9. Liew, Hoon-Voon, and Lim, Teik C. (2005): “Analysis of Time-Varying Rolling Element Bearing Characteristics,” *J. of Sound and Vibration*, vol. 283, pp. 1163–1179.
10. Joshi, Prasad Raman (2004): “An Elastic Contact Theory for Modeling Vibration Transmissibility through Rolling Element Bearing,” MS Thesis, University of Cincinnati.
11. Royston, T.J., and Basdogan, I. (1998): “Vibration Transmission through Self-Aligning (Spherical) Rolling Element Bearings: Theory and Experiment,” *J. of Sound and Vibration*, vol. 215, no. 5, pp. 997–1014.
12. Lewis, P., and Malanoski, S. B. (1965): “Rotor-Bearing Dynamics Design Technology; Part IV: Ball Bearing Design Data,” AFAPL–TR–65–45, Part IV.
13. Zeillinger, Robert, and Köttritsch, Hubert (1996): :Damping in a Rolling Bearing Arrangement,” *Evolution*, issue 1/96.
14. Fleming, David P. (2002):Effect of Bearing Dynamic Stiffness on Gear Vibration. Proceedings of Ninth International Symposium on Transport Phenomena and Dynamics of Rotating Machinery (ISROMAC-9), Yoshinobu Tsujimoto, ed. Pacific Center of Thermal-Fluids Engineering, Kihei, Maui, Hawaii, U.S.A.
15. DyRoBeS, version 11.3 (2006), Eigen Technologies Inc., http://www.DyRoBeS.com/
16. Fleming, David P. (1996): “Rotordynamics on the PC: Transient Analysis with ARDS,” *Rotating Machinery - 1996* (Proceedings of the Sixth International Symposium on Transport Phenomena and Dynamics of Rotating Machinery), Vol. I, D.C. Han, S.T. Ro, and J.H. Kim, eds., Pacific Center of Thermal-Fluids Engineering., Kihei, Maui, Hawaii, U.S.A., pp. 367–375.
17. Dimofte, Florin; Ene, Nicoleta; Keith, Theo G. Jr.; and Handschuh, Robert F. (2006): “Investigation of the Stability of a Rotor Supported by Oil Journal Wave Bearing,” Proceedings of the IFToMM Seventh International Conference on Rotor Dynamics, H. Springer and H Ecker, eds., Technical University of Vienna, Vienna, Austria.
18. Dimofte, Florin, University of Toledo (2007): Personal communication.

REPORT DOCUMENTATION PAGE			*Form Approved* *OMB No. 0704-0188*

1. REPORT DATE *(DD-MM-YYYY)*
01-07-2007

2. REPORT TYPE
Technical Memorandum

3. DATES COVERED *(From - To)*

4. TITLE AND SUBTITLE
Vibration Transmission Through Bearings With Application to Gearboxes

5a. CONTRACT NUMBER

5b. GRANT NUMBER

5c. PROGRAM ELEMENT NUMBER

6. AUTHOR(S)
Fleming, David, P.

5d. PROJECT NUMBER

5e. TASK NUMBER

5f. WORK UNIT NUMBER
WBS 877868.02.07.03.04.01

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)
National Aeronautics and Space Administration
John H. Glenn Research Center at Lewis Field
Cleveland, Ohio 44135-3191

8. PERFORMING ORGANIZATION REPORT NUMBER
E-16131

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)
National Aeronautics and Space Administration
Washington, DC 20546-0001

10. SPONSORING/MONITORS ACRONYM(S)
NASA

11. SPONSORING/MONITORING REPORT NUMBER
NASA/TM-2007-214954; ISCORMA-4-510

12. DISTRIBUTION/AVAILABILITY STATEMENT
Unclassified-Unlimited
Subject Category: 37
Available electronically at http://gltrs.grc.nasa.gov
This publication is available from the NASA Center for AeroSpace Information, 301-621-0390

13. SUPPLEMENTARY NOTES

14. ABSTRACT
Cabin noise has become a major concern to manufacturers and users of helicopters. Gear noise is the largest part of this unwanted sound. The crucial noise path is generally considered to be from the gears through the gear-supporting shafts and bearings into the gearbox case, and from there either through the gearbox mounts or the surrounding air to the helicopter cabin. If the noise, that is, the gear and shaft vibration, can be prevented from traveling through the gearbox bearings, then the noise cannot make its way into the helicopter cabin. Thus the vibration-transmitting properties of bearings are of paramount importance. This paper surveys the literature concerning evaluation of properties for the types of bearings used in helicopter gearboxes. A simple model is proposed to evaluate vibration transmission, using measured or calculated bearing stiffness and damping. Less-commonly used types of gearbox bearings (e.g., fluid film) are evaluated for their potential in reducing vibration transmission.

15. SUBJECT TERMS
Gear noise; Vibration transmission; Bearing properties

16. SECURITY CLASSIFICATION OF:			17. LIMITATION OF ABSTRACT	18. NUMBER OF PAGES	19a. NAME OF RESPONSIBLE PERSON
a. REPORT	b. ABSTRACT	c. THIS PAGE			STI Help Desk
U	U	U		13	19b. TELEPHONE NUMBER *(include area code)* 301-621-0390